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Analysis of ratio influence of discharge and suction pressure on operating processes on hybrid power piston volumetric machines

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Abstract

The paper considers the prospective design of hybrid power piston volumetric machines through the use of gas pressure oscillations in the discharge line. Using the developed mathematical model, a numerical experiment of the effect of the ratio of discharge pressure and the suction pressure on the operating processes, power and loss characteristics of the hybrid power piston volumetric machines

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Keywords:compressor; operating processes; cooling; hybrid machine.

1. Introduction

Considered impact on the performance and efficiency of the compressor operation, as well as the conditions for its safe operation is due to cooling the compressed gas and the elements of the working chamber of the compressor. When gas is compressed in compressors of volumetric action temperature and pressure of compressed gas are substantially increased. Increasing the gas pressure increases the load acting on the working bodies of the compressor and distribution systems (valves). An increase in temperature leads to a deterioration in economic performance of the compressor.

Currently, to improve the efficiency of the piston compressors they combine their work and piston pumps work [1, 2]. This unit is named a hybrid piston power machine. These machines increased gas compression efficiency with improving the cooling of compressed gas, reducing leaks and reducing the work of friction forces. In addition the machine can be used in the petrochemical industry.

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2. Study subject

The authors of the papers [3] have proposed the design of the perspective piston power hybrid volumetric machine through the use of pressure fluctuations in the gas discharge line; its schematic diagram is shown in Fig. 1.

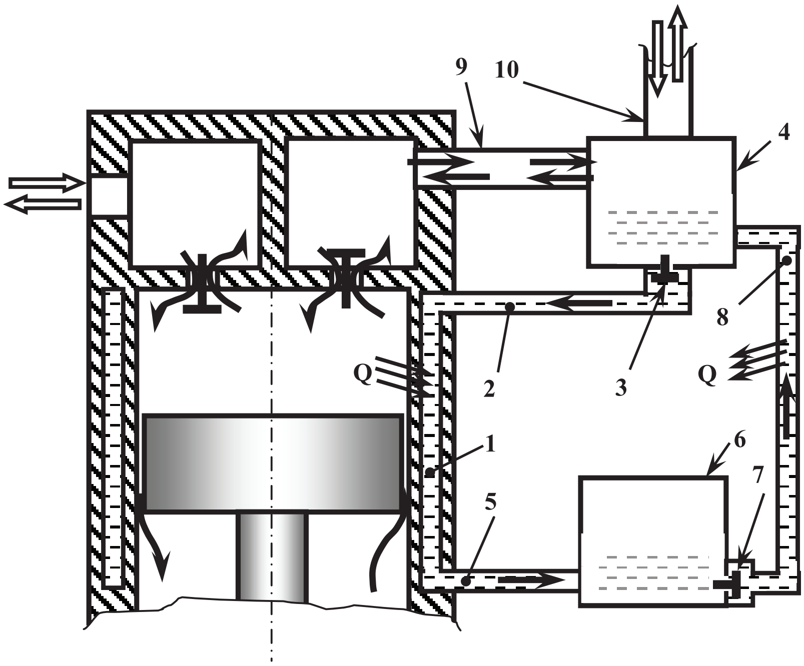

Fig. 1. Schematic diagram of a hybrid power piston machines using gas pressure fluctuations in the discharge line.

The operating principle of the illustrated construction is that compressed gas from the working chamber comes into discharge cavity through the discharge valve. From the discharge cavity the gas enters receiver 4 through pipeline 1. The lower part of receiver 4 is filled with liquid and the upper part is filled with gas. Pipeline 2 connects the lower part of receiver 4 through a check valve with jacket space 3 of the compressor cylinder. The lower part of the jacket space is connected to the pipeline and the lower part of receiver 6, which is also filled with liquid as well as the lower part of receiver 4. The upper part of receiver 6 is filled with gas. The lower part of receiver 6 is also connected through check valve and pipeline 8 with the lower part of receiver 4. From the upper part of receiver 4 the gas is supplied to the consumer through the regulating valve.

3. Methods

To solve effectively the problems in the development and study of the proposed design of the piston hybrid power machine on the basis of modern methods of mathematical stimulating of pump compressors operating processes [4,5] there was developed a mathematical model of piston hybrid power machine operating processes. The mathematical model of operating processes includes the calculation of thermodynamic parameters in the chambers of constant and variable volume, the calculation of non-stationary one-dimensional motion of gas in connecting pipeline 1, unsteady motion of fluid in the connecting pipelines. The mathematical model of operating processes is based on fundamental laws of conservation of mass, movement and energy, but also the ideal gas equation is used.

Calculation of non-stationary one-dimensional motion of gas was conducted with "coarse particles" method. Systems of ordinary differential equations were solved with Euler's method.

The adequacy of the developed mathematical model is confirmed by the use of the fundamental laws of conservation of mass, movement and energy, as well as a pilot study conducted [3].

4. Results and discussion

The developed mathematical model allows analyzing the impact of major design parts and mode parameters on the economic and operability of the piston hybrid power machine.

As a base case for the analysis we will take PGEMOD having the main structural dimensions shown in Table 1 and Fig. 2.

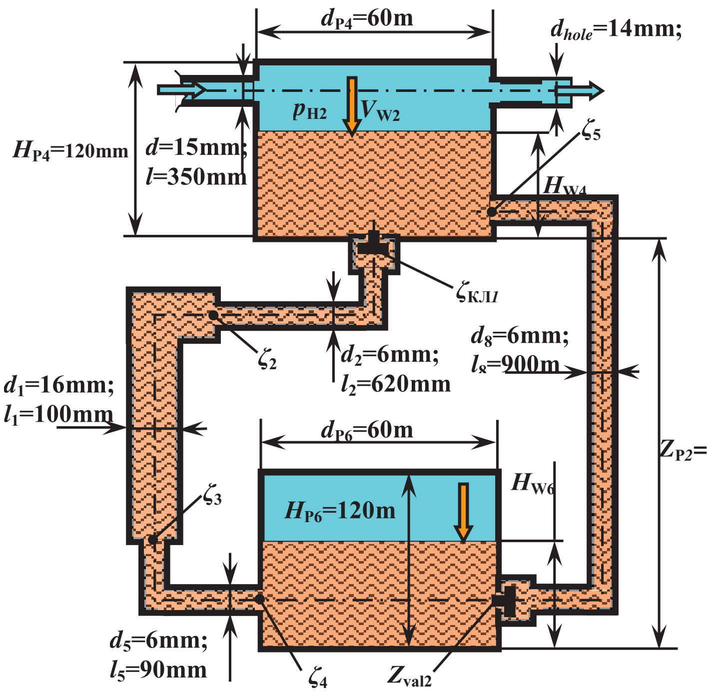

Fig. 2. Schematic diagram of the pump section of the base variant PGEMOD showing the main geometric dimensions.

Table 1.Main geometric dimensions of PGEMOD.

PGEMOD element	Characteristic
Gas part of the experimental sample of PGEMOD	
Cylinder	Cylinder diameter – 42mm
	The piston stroke length – 38mm
	The length of the cooling jacket – 100mm
	The outer diameter of the jacket– 58mm
	The inner diameter of the jacket– 48mm

	Linear end volume – 3mm
Suction gas valve	Seat diameter – 15mm The diameter of the plate – 17mm Stroke – 1.5mm The stiffness of the spring – 4.23N/mm Preload – 0mm Weight of the valve closure – 2.5g
Discharge gas valve	Seat diameter – 10mm The diameter of the plate – 12mm Stroke – 1.5mm The stiffness of the spring – 4.7N/mm Preload – 0mm Weight of the valve closure – 5g Number of valves – 2 pcs
Cylinder head	The amount of suction – 20 cm^3 The volume on the discharge – 28 cm^3
Suction pipeline	The inner diameter – 12mm Length – 35mm
The gas pipeline from the compressor to receiver 4	The inner diameter – 8mm Length – 550mm
Gas pipeline from receiver 4 to receiver buffer	The inner diameter – 14mm Length – 1500mm
Receiver buffer	Volume – 25 l
The liquid part of the experimental sample of PGEMOD	
Receiver 4	Cylinder diameter – 60mm Cylinder height – 120mm Fluid or gas inlet diameter – 16mm
Receiver 6	Cylinder diameter – 60mm Cylinder height – 120mm Fluid inlet diameter – 16mm
Fluid conductor from the cooling jacket to receiver 4	The inner diameter – 6mm Length – 290mm
Fluid conductor from receiver 4 to receiver 6	The inner diameter – 6mm Length – 335mm
Fluid conductor from receiver 6 to the cooling jacket	The inner diameter – 6mm Length – 285mm
Liquid valve	Seat diameter – 12mm The diameter of the plate – 14mm Stroke– 4mm The stiffness of the spring – 4g/mm Preload – 3mm

Weight of valve closure – 2.4g

As a response function that determines the efficiency of PGEMOD, we take the following parameters:

1. The coefficient of flow compressor section is defined as:

$$\lambda = \frac{M_p}{V_h \rho_s} \tag{1}$$

where

$$M_p = \oint (dM_7 - dM_8)$$

V_h is volume described with the piston;

$\rho_s = \frac{p_s}{RT_s}$ is density of gas suction.

2. Indicative isothermal efficiency:

$$\eta_{ind.is.} = \frac{\oint V dp}{A_{ind.is}} \tag{2}$$

where

$$A_{ind.is} = M_p R T_s \ln\left(p_d / p_s \right)$$

T_s is suction temperature.

3. The relative operational losses in the process of absorption:

$$\frac{\Delta A_{вс}}{A_ц} = \frac{\int_{V_1}^{V_h+V_m} (p_{вс} - p_c) dV}{\oint p_c dV} \tag{3}$$

where $\boldsymbol{V_1}$ is the volume of the working chamber, corresponding to the beginning of the process of absorption.

4. The relative loss of operation during discharge process:

$$\frac{\Delta A_d}{A_c} = \frac{\int_{V_2}^{V_h} (p_c - p_d) dV}{\oint p_c dV} \tag{4}$$

where $\boldsymbol{V_2}$ is the volume of the working chamber of the compressor section, corresponding to the beginning of the injection process.

5. Relative pressure losses during discharge process:

$$\frac{\Delta p_d}{p_d} = \frac{\int_{V_2}^{V_e} (p_c - p_d) dV \Big/ (V_2 - V_e)}{p_d} \tag{5}$$

6. The relative loss of pressure in the suction process:

$$\frac{\Delta p_s}{p_s} = \frac{\int_{V_2}^{V_h + V_e} (p_s - p_c) dV \Big/ (V_h + V_e - V_1)}{p_s} \tag{6}$$

7. The ratio of the maximum pressure in the pumping chamber to the minimum:

$$\frac{P_{d1\max}}{P_{d1\min}} \tag{7}$$

and the value of the relative pressure change:

$$\delta_{pd1} = \frac{p_{d1\max} - p_{d1\min}}{p_{d1\max}} \tag{8}$$

8. The ratio of the maximum pressure in the gas chamber of receiver 4 to the minimum:

$$\frac{P_{d2\max}}{P_{d2\min}} \tag{9}$$

and the value of the relative pressure change:

$$\delta_{pd2} = \frac{p_{d2\max} - p_{d2\min}}{p_{d2\max}} \tag{10}$$

9. The ratio of maximum pressure in the gas chamber of receiver 6 to minimum:

$$\frac{P_{d3\max}}{P_{d3\min}} \tag{11}$$

and the value of the relative pressure change:

$$\delta_{pd3} = \frac{p_{d3\max} - p_{d3\min}}{p_{d3\max}} \tag{12}$$

10. The average speed of the fluid in receiver 4 in the cycle:

$$V_{2wav} = \frac{\sum_{i=1}^{j_1} \int_0^{\Delta\varphi_i} V_{2w} d\varphi}{\sum_{i=1}^{j} \Delta\varphi_i} \tag{13}$$

where j_1 is the number of intervals $\Delta\varphi_i$ in the cycle at which speed V_{2w} is above zero;

$\Delta\varphi_i$ is the value of the rotation angle interval, in which V_{2w} is above zero.

11. The average speed of the fluid in receiver 6 in the cycle:

$$V_{3wav} = \frac{\sum_{i=1}^{j_2} \int_0^{\Delta\varphi_i} V_{3w} d\varphi}{\sum_{i=1}^{j} \Delta\varphi_i} \tag{14}$$

where j_2 is the number of intervals $\Delta\varphi_i$ in the cycle at which speed V_{3w} is above zero;

12. The ratio of the masses of coolant flows through the cooling jacket of receiver 4 to receiver 6 for the cycle M_w to the mass of gas supplied to the consumer M_g:

$$\frac{M_w}{M_g} = \delta_w \tag{15}$$

13. Uneven gas supply to the consumer:

$$\delta M_g = \frac{\Delta M_{2\max} - \Delta M_{2\min}}{\Delta M_{2\max}} \tag{16}$$

where $\Delta M_{2\max}$ is the maximum value of elementary mass of gas supplied to the consumer during $\Delta\tau$;

$\Delta M_{2\min}$ is the minimum value of elementary mass of gas supplied to the consumer during $\Delta\tau$.

One of the basic mode parameters of the compressor and the volumetric pump is a ratio of discharge pressure to suction pressure $\varepsilon = p_d / p_s$. Air compressors of household and general purpose have suction pressure equal to 0.1 MPa, and the discharge pressure varies between 0.3 and 0.8 MPa respectively ε ranges from 3 to 8. As a result we consider the range of PGEMOD operation at the range ε from3 to 8, with the most typical enginespeed $n_{rev} = 1500 rev/m$.

With increasing discharge pressure and thus with increasing ε gas compression process is increased, and the discharge process decreases; the process of reverse expansion increases and absorption process decreases. Reducing gas absorption process and a number of other reasons, such as increasing the heating up at suction, increasing the amount of compressed gas leakage etc. [4] reduces compressor section supplying ratio (see Fig. 3)

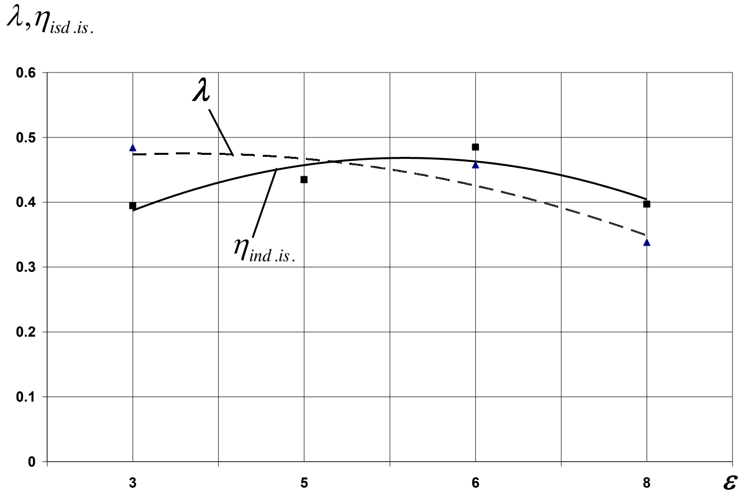

Fig. 3. Dependence of factor of supply and indicator of isothermal efficiency on ε .

Reducing the duration of the discharge process and also increasing its density leads to the fact that the amplitude of gas oscillations in the chamber increases (Fig. 4). Value $\frac{p_{d1\max}}{p_{d1\min}}$ at $\varepsilon = 3$ is $\frac{366922}{287125}$, and if $\varepsilon = 8$ the same value is $\frac{859765}{758919}$, i.e. the difference between the maximum and the minimum pressure is almost $10^5\ Pa$ (1 atmosphere). However, this difference referred to the maximum pressure δ_{pd1} decreases (see Fig. 5). Here are marked the values δ_{pd2} and δ_{pd3} in the function from ε. Values δ_{pd2} and δ_{pd3} also decrease, although the amplitude of the pressure oscillation in receiver 4 and in receiver 4 are increased (see Fig. 6). The results presented in Fig. 6 show that the difference between the pressures p_{d3} and p_{d2} on the section $3.1415 \le \varphi \le 5.85$ variable sign, although if $\varepsilon = 3$ this difference is always positive.

Increasing the flow pulsations increases the velocity fluctuations V_{2w} and V_{3w} (see Fig. 7), and this in turn leads to a decrease in average velocity V_{2wav} and V_{3wav} (see Fig. 8) and a decrease in the coolant flow supplied to the cooling section of the compressor.

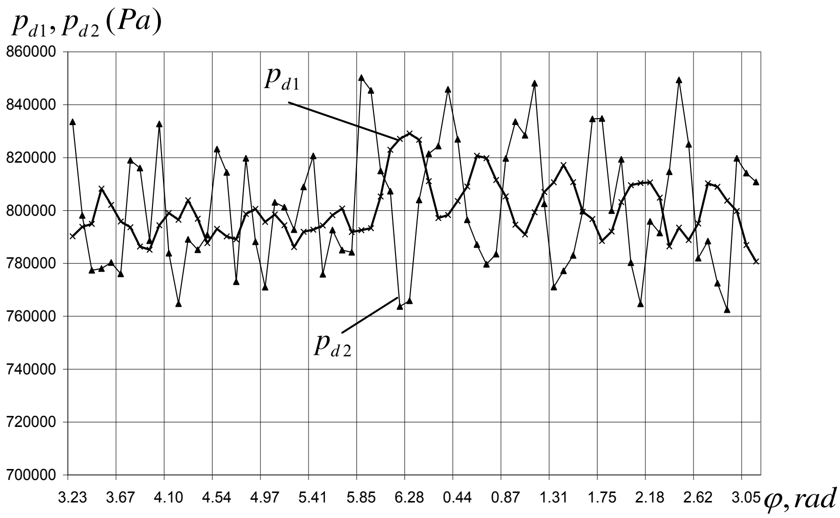

Fig.4. The change dependence of the instantaneous pressures in the pressure chamber and in receiver 4 from the crank angle.

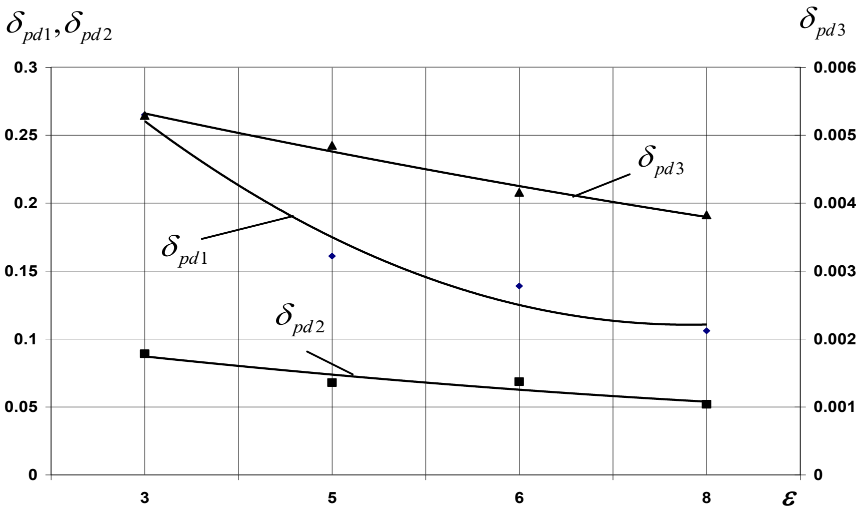

Fig. 5. Dependence of the relative changes in pressure in the discharge chamber and receivers from ε .

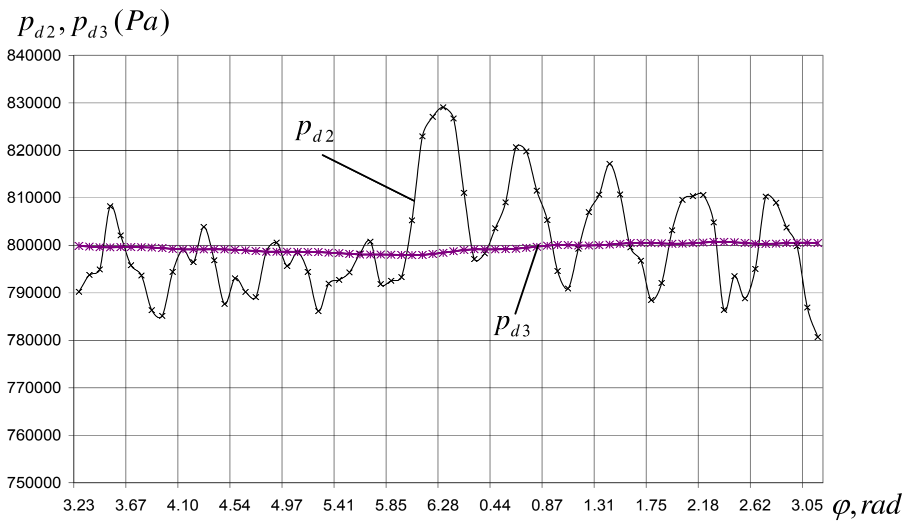

Fig. 6. Dependence of instant pressures in the receiver chambers from the crank angle.

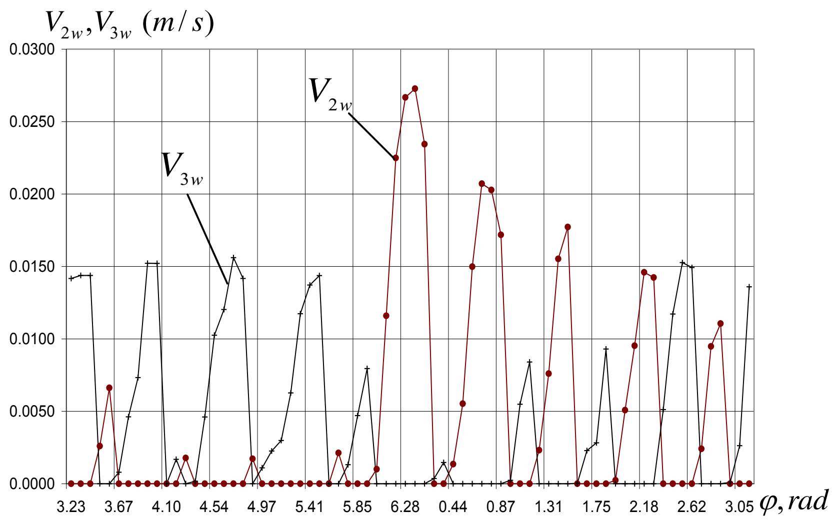

Fig. 7. Dependence of instantaneous fluid velocities in the receivers on the crank shaft.

5. Conclusion

Analyzing the results presented in Fig. 3 - 7 one can make, except for the above, some additional findings:

1. Max speed V_{2w} at $\varepsilon = 8$ is slightly higher than the maximum speed V_{2w} at $\varepsilon = 3$.
2. The average speed V_{2wav} and V_{3wav} with increasing ε decreases almost linearly.
3. Reducing the average speed V_{2wav} or V_{3wav} reduces the flow of coolant $\frac{(G_{w1} + G_{w2})}{2}$.
4. The value of the average velocity V_{3wav} is less than the average speed V_{2wav}, although values G_{w1} and G_{w2} are approximately equal. This is because the time interval or the angle at which they are no longer to zero speed V_{2w} is less than for speed V_{3w}.

With increasing magnitude ε the amount of coolant G_w, decreases as well as the amount of gas supplied to the consumer. But due to the fact that the reduction of gas flow is faster than the decrease in the coolant flow, the ratio G_w/G_g increases (see Fig. 9).

With increasing ε fluctuations in gas supply to consumers increase. With increasing ε from 3 to 8 one increased δ_d and it increased almost doubled from 0.8 to 1.61.

With increasing ε one decreases relative losses and working pressure in the suction and discharge (see Fig.10). Reducing the relative operation losses leads to an increase in the indicator of isothermal efficiency.

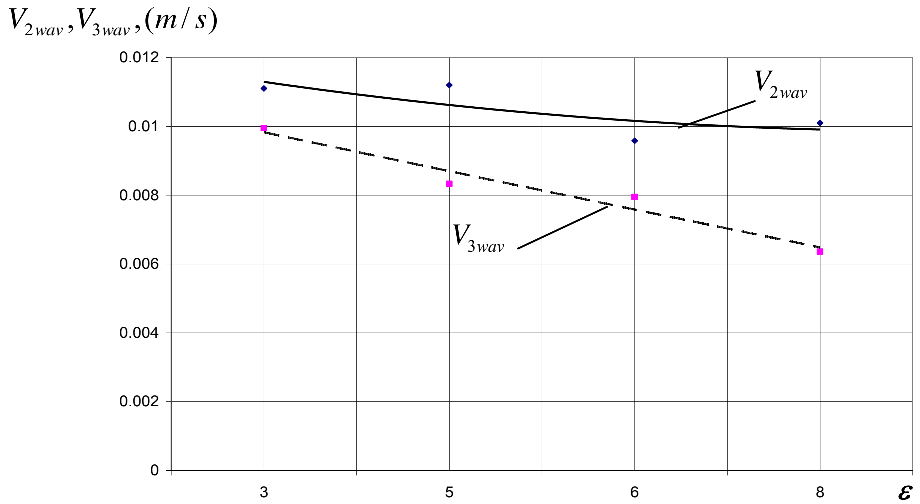

Fig. 8. Dependence of the average velocity of the fluid in the receivers on ε.

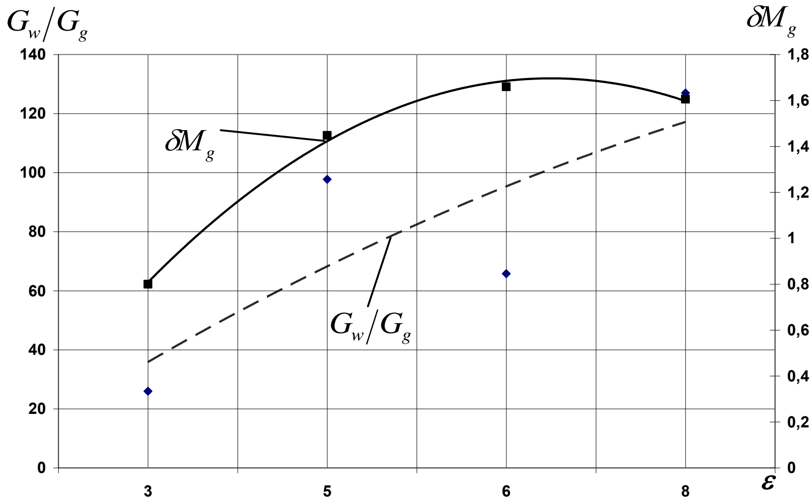

Fig. 9. Dependence of the relative coolant flow rate and non-uniformity of gas supply to the consumer on ε .

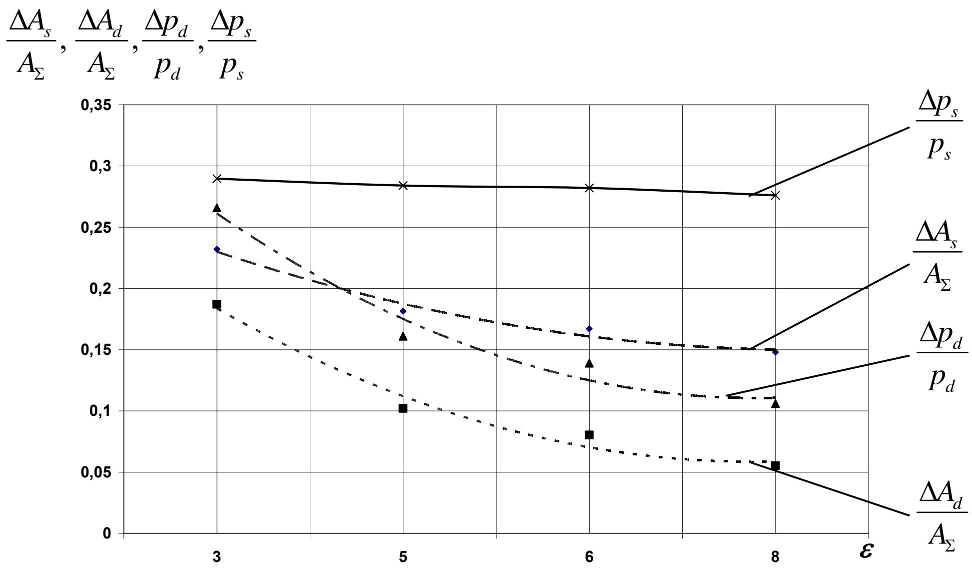

Fig. 10. Dependences of relative pressure loss and operation in the suction and discharge processes on ε .

With further increase in ε expenditure of operation during gas compression in comparison with isothermal compression increases, resulting in decreased isothermal indicative efficiency (see Fig. 3). Maximum value $\eta_{ind.is}$ is achieved at $\varepsilon = 6$.

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References

[1] V.Ye. Shcherba, A.P. Bolshtyansky, V.V. Shalai, A.V. Hodoreva Pump compressors. Working processes and principles of design, M: Mechanical engineering, 2013, 388 p. (in Russian)
[2] V.Ye. Shcherba, A.K. Kuzhbanov, G.A. Nesterenko, Calculation of the compression process and the expansion piston pump compressor with gas damping, Omsk scientific bulletin, Omsk: Publishing House of the Omsk State Technical University, 2012, № 23 (110), pp. 148-152. (in Russian)
[3] I.E. Lobov, Design and experimental study of the cooling system of a piston compressor, based on the oscillation phenomena in gas discharge line, Omsk Scientific Bulletin. Ser. Devices, machines and technology, 2015, № 3 (143), pp.145-149. (in Russian)
[4] P.I. Plastinin, Reciprocating compressors. Theory and evaluation. Volume 1, M .: Kolos, 2000, 456 p. (in Russian)
[5] V.Ye. Shcherba, Workflows of volumetric compressor, M .: Nauka, 2008, 319 p. (in Russian)